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By John W. R. Creagh, Melvin J. Hartmann
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Cleveland, Ohio

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SUMMARY

An investigation was conducted to determine the effect on the performance of a two-valve air-cooled aircraft cylinder of increasing the valve overlap from the conventional value of 40° to 130°. Direct cylinder injection of fuel was used throughout the investigation, which covered a range of inlet and exhaust pressures at engine speeds of 2000, 2500, and 2800 rpm. During some of the runs with 130° valve overlap, a gas-sampling valve was installed in the cylinder head to determine the combustion fuel-air ratio.

With the 130° valve overlap, nearly complete clearance-volume scavenging was obtained with a consequent increase in power output of approximately 20 percent over that obtained with no scavenging. Scavenging without loss of fresh charge was confined to a narrow range of exhaust-to-inlet-pressure ratios, especially at the lower engine speeds. Gas samples taken from the cylinder head during the combustion process showed that, when complete clearance-volume scavenging was obtained, approximately 50 percent of the increased charge-air flow was wasted in the scavenging process. Despite the increased power output with the 130° valve overlap, the cooling effect of the scavenging air was sufficient to prevent any increase in the exhaust-valve-seat temperature and decreased the exhaust-valve-guide temperature as much as 50° F.

INTRODUCTION

The use of large valve overlap for scavenging and charging the cylinders of a supercharged engine has been shown to be an effective means of increasing the power. Schey and Young (reference 1) have shown that for low-speed operation with a four-valve cylinder practically all the ideal increase in power obtainable with complete scavenging was realized with a valve overlap of 112°. Slightly longer periods of overlap were used in the investigation of reference 2 to obtain comparable scavenging at intermediate speeds and it was shown that scavenging a two-valve combustion chamber was more difficult than scavenging a four-valve chamber. The difference was attributed to air flowing directly from the inlet port to the exhaust port of the two-valve cylinder without completely sweeping the residual gases from the combustion chamber. This condition may be aggravated if a low-turbulence condition exists.

Valve-overlap scavenging in combination with direct fuel injection and an exhaust turbine is considered a good method of increasing the net power of an engine. Within a limited range of altitude, valve-overlap scavenging is beneficial without the use of an exhaust turbine. Although the general performance effects of valve-overlap scavenging have been known, the actual performance characteristics over a wide range of exhaust-to-inlet-pressure ratios have been unavailable.

An investigation was made at the NACA Cleveland laboratory to determine, for a two-valve air-cooled aircraft cylinder, the effects of large valve overlap on performance over a wide range of exhaust and inlet-manifold pressures. The single-cylinder engine performance furnishes cylinder cycle information necessary for analysis of a combination of engine, supercharger, and exhaust turbine. The valve overlap used in this investigation, which was selected from references 1 and 2, was 130°. Comparative performance was obtained with 40° overlap, which is standard timing for the engine from which the cylinder was taken. The performance of the engine with the two valve timings was compared over a range of exhaust-to-inlet-pressure ratios from approximately 0.1 to 1.3 at engine speeds of 2000, 2500, and 2800 rpm. During part of the investigation with 130° overlap, the combustion fuel-air ratio was determined from gas samples obtained through a sampling valve installed in the cylinder head. From these data, the relative magnitudes of charge air burned and charge air wasted or lost through the exhaust valve during the scavenging process could be determined. The internal-cooling effect of large scavenging air flow was measured by comparing data on internal cylinder temperatures for similar operating conditions with both valve timings at a constant rear-spark-plug-gasket temperature.

APPARATUS AND INSTRUMENTATION

Engine. - This investigation was conducted on a single-cylinder engine consisting of a two-valve, air-cooled, aircraft cylinder mounted on a CUE crankcase. A schematic diagram of the setup is shown in figure 1. The cylinder had a bore of $6\frac{1}{8}$ inches and a stroke of $6\frac{7}{8}$ inches with a measured compression ratio of 6.1. An NACA balanced-diaphragm torque indicator (reference 3) was used to measure the power absorbed or delivered by a 300-horsepower dynamometer. The speed of the dynamometer was controlled by an electronic device. Engine speed was measured by an electrically interconnected stop watch and revolution counter. The spark timing was set at 20° B.T.C. The over-all accuracy of the instrumentation was estimated to be between 1.0 and 1.5 percent.

Valve timing. - Standard valve timing for the cylinder produced 40° valve overlap. Because increasing the valve-overlap period to 130° without changing the inlet-closing and the exhaust-opening periods was desired, a set of cams was constructed by inserting suitable dwells at the maximum lift portion of the 40°-overlap cam profiles, thereby maintaining the same rate of valve travel and maximum valve lift for both sets of cams. The two valve timings were:

	Valve-overlap cams	
	40°	130°
Inlet opens	22° B.T.C.	66° B.T.C.
Inlet closes	46° A.B.C.	46° A.B.C.
Exhaust opens	80° B.B.C.	80° B.B.C.
Exhaust closes	18° A.T.C.	64° A.T.C.

Fuel system. - A sketch of the high-pressure part of the fuel system is shown in figure 2. Fuel was supplied from a single-cylinder high-speed pump that held the start of injection constant and allowed the end of injection to vary with fuel flow. The fuel was directly injected into the cylinder with a spring-loaded nozzle at a pressure of 400 pounds per square inch. The nozzle was located between the front spark plug and the inlet valve and sprayed fuel across the path of the incoming charge air. The fuel used throughout the procedure was 100-octane gasoline and was measured with a 200 pound-per-hour rotameter calibrated for this fuel.

Cooling-air system. - A steel duct encased the cylinder and a set of standard baffles directed the cooling air around the cylinder. The baffles were extended to seal the sides, the top, and the bottom of the steel duct, as shown in figure 2. The cooling air was taken from the laboratory cooling-air system, brought to the front of the steel duct through a 16-inch pipe, removed from the rear of the duct through a similar pipe, and discharged to the atmosphere.

Charge-air and exhaust systems. - Charge air was supplied from the laboratory combustion-air system through a pressure-control valve and a set of electrical air heaters to a surge tank of about 10-cubic-feet capacity, which dampened pulsations in the system. The inlet pipe and the surge tank (fig. 2) were insulated to prevent excessive heat losses. The inlet-manifold-pressure tap was located in the surge tank. Charge-air weight was measured with a 1.5-inch thin-plate orifice in a pipe of 4.02-inch inside diameter located between the pressure-control valve and the air heater. All measurements were taken to comply with A.S.M.E. orifice practice for a thin-plate orifice and flange pressure taps.

As shown in figure 1, the exhaust gases from the engine flowed into a 4-cubic-foot muffler. From the muffler, the exhaust gases flowed into a cooling tank of about 5-cubic-feet capacity and then to the laboratory exhaust systems. The pressure in the exhaust system was measured near the tank. Cooling water was sprayed into the exhaust pipe about 28 inches downstream of the cylinder exhaust flange.

Gas-sampling equipment. - In order to determine the actual operating fuel-air ratio for some of the runs with the 130° valve overlap, exhaust-gas samples were taken from the combustion chamber with an electrically operated sampling valve. The sampling element was fitted with electrodes and served both as a front spark plug and as a sampling valve. The valve in its normal position was held closed by a spring and was opened by a magnetic coil that obtained current from a condenser. The condenser was discharged by a set of contact points that were fitted with a vernier adjustment to compensate for speed. The valve was set to open at 100° A.T.C. on the power stroke and was estimated to stay open for crank travel of 2° to 3°, depending on the engine speed. The gas samples were passed through an oxidizing furnace and then analyzed in an Orsat apparatus according to the method of reference 4.

Temperature measurements. - Probe-type thermocouples were used to measure the oil and charge-air temperatures. Temperatures of the exhaust-valve guide and seat were measured by a thermocouple inserted in each of these parts. The temperature under the rear spark plug was measured with a gasket-type thermocouple. All temperatures were read on a self-balancing potentiometer.

METHODS AND PROCEDURE

Preliminary investigations. - The fuel pump was timed to start injecting fuel into the cylinder at 70° A.T.C. in accordance with the results of reference 2. The timing was visually set through the use of a stroboscopic light. Investigations were made over a range of engine speed and fuel flow, for which the start of injection did not vary more than ±5 crank degrees throughout. This accuracy was considered satisfactory because the engine performance was insensitive to slight changes in injection timing in this part of the cycle.

The accuracy of the exhaust-gas sampling valve was determined by operating the engine with the 40°-overlap cams at equal inlet-manifold and exhaust pressures over a range of fuel-air ratios at an engine speed of 2000 rpm. The results of the investigation are shown in figure 3 where the fuel-air ratios determined from the measured quantities of charge-air and fuel flow are compared with the fuel-air ratios determined from the exhaust-gas sampling valve. Close agreement between the two methods of determining the fuel-air ratio was obtained. The percentage change in indicated mean effective pressure from the maximum obtained as the fuel-air ratio was varied from approximately 0.05 to 0.105 is also shown in figure 3. Maximum indicated mean effective pressure was obtained at a fuel-air ratio of 0.08; the percentage change in power with an incremental change in fuel-air ratio was much smaller on the rich side of 0.08 than on the lean side.

With standard valve timing, the overlap period was assumed to be so small that no fresh charge air would be lost through the exhaust valve under any of the operating conditions of the program. In order to prove this assumption, an investigation was conducted in which the inlet-manifold pressure and temperature, the engine speed, and the fuel-air ratio were held constant at 35 inches of mercury absolute, 200° F, 2000 rpm, and 0.09, respectively, and the exhaust pressure was varied from approximately inlet-manifold pressure to a value considerably below it. The gas-sampling valve was used to determine the combustion fuel-air ratio, and because the fuel was injected after the exhaust valve closed, any loss in charge air would be indicated by a difference between the fuel-air ratios determined from the measured intake of fuel and air and from the gas-sampling valve. The following table shows the results of the investigation:

Exhaust pressure, Inlet pressure p_e/p_m	Fuel-air ratio, measured intake	Fuel-air ratio, exhaust-gas sampling valve
1.077	0.0915	0.0912
.994	.0909	.0902
.811	.0909	.0906
.759	.0906	.0910
.434	.0906	.0916
.243	.0906	.0915

These data show that a negligible amount of charge air was lost inasmuch as the two methods of measuring fuel-air ratio indicate nearly the same values. It was therefore considered safe to eliminate the use of the gas-sampling valve during operation with the 40° overlap.

Performance investigations. - The engine was operated at speeds of 2000 and 2500 rpm with inlet-manifold pressures of 25, 35, and 45 inches of mercury absolute and at 2800 rpm with inlet-manifold pressures of 30 and 40 inches of mercury absolute. At each engine speed and inlet manifold pressure, the exhaust pressure was varied over a wide range for both the 40° and 130° valve-overlap timings. All investigations were run with a rear-spark-plug-gasket temperature of 400° F and a charge-air temperature of 200° F. The friction horsepower was obtained by motoring the engine at the same inlet-manifold pressure, exhaust pressure, and speed as the power runs.

Because it was shown in the preliminary investigations that no charge air was lost through the exhaust valve with 40° valve overlap, the ratio of fuel flow to charge-air flow was considered as the combustion fuel-air ratio and was held constant at a value of 0.09. During those runs with the 130° overlap in which the fuel-air ratio was not determined by using the gas-sampling valve, the fuel flow was set by maintaining the same indicated specific fuel consumption as had been previously obtained under the same conditions with the 40° overlap valve timing. Occasional checks with the gas-sampling valve showed that this method of setting the fuel flow was sufficiently accurate to warrant its use for mixture setting and also saved considerable time. In operation with exhaust pressures higher than inlet pressures, the fuel-air ratio with the 130° valve overlap timing could, of course, be determined from the measured inlet values because no charge air was wasted.

RESULTS AND DISCUSSION

General performance characteristics. - The curves of figure 4 show the effect on indicated mean effective pressure and charge-air flow of varying the ratio of exhaust absolute to inlet absolute pressure p_e/p_m for both 40° and 130° valve overlaps. A much greater variation in both indicated mean effective pressure and charge-air flow was obtained with the 130° overlap. When the exhaust pressure was lower than the inlet pressure, the increase in indicated mean effective pressure with the 130° overlap was largely due to the action of the scavenging air in sweeping the residual gases from the clearance volume of the cylinder. The increase in charge-air flow was due both to the increased air burned in the clearance space as a result of scavenging and to air lost through the exhaust valve during the scavenging process. As the exhaust pressure increased above the value of inlet pressure, the charge-air flow and indicated mean effective pressure rapidly decreased owing to the presence of excessive exhaust gas in the cylinder resulting from reversal of flow through the exhaust valve. The comparatively small change in both indicated mean effective pressure and charge-air flow with 40° valve overlap can be attributed to the expansion or compression of the residual gases in the clearance volume of the cylinder resulting when the exhaust pressures are higher or lower than the inlet pressure.

The following table shows the maximum percentage increase in indicated mean effective pressure with an increase in the valve overlap obtained at any value of p_e/p_m over the range of engine speeds and manifold pressures covered by this investigation.

Engine speed (rpm)	Inlet manifold pressure (in. Hg abs.)	$\frac{\text{Exhaust pressure}}{\text{Inlet pressure}}$, p_e/p_m	imep with 130° valve overlap (lb/sq in.)	imep with 40° valve overlap (lb/sq in.)	Increase in imep with 130° valve overlap (percent)
2000	25	0.77	143.5	124.5	15.3
2000	35	.77	201.0	176.5	13.9
2000	45	.70	264.0	232.0	13.8
2500	25	.70	138.5	118.5	16.9
2500	35	.75	191.0	169.0	13.0
2500	45	.75	247.0	220.5	12.0
2800	30	.82	155.0	136.0	14.0
2800	40	.75	206.5	184.0	12.2

The value of p_e/p_m at which the maximum percentage increase in indicated mean effective pressure occurred in the preceding table need not and sometimes does not coincide with the value of p_e/p_m corresponding to maximum power output with 130° valve overlap because in this range of p_e/p_m, the indicated mean effective pressure was increasing with decreasing p_e/p_m for both the 40° and 130° overlaps. The increased indicated mean effective pressure shown in this table is that obtained with the increased scavenging effect of the higher valve overlap. Nearly all the maximum percentage increases in power were obtained at values of p_e/p_m between 0.70 and approximately 0.80, irrespective of engine speed or manifold pressure.

Because other investigations have experienced more difficulty in scavenging two-valve cylinders than the several other types investigated, it is interesting to determine whether complete scavenging was obtained. As mentioned in reference 1, the theoretical ratio of power with complete clearance-volume scavenging to power with no scavenging is $r/r-1$, where r is the compression ratio. Owing to probable inertia effects, the power curves of figure 4 do not intersect when the exhaust and inlet pressures are equal. The indicated mean effective pressure value at which no clearance-volume scavenging occurred probably should be taken as the point on the 40°-overlap curves corresponding to p_e/p_m value of 1.0. The following table shows the ratio of maximum power with 130° overlap to power with no clearance-volume scavenging, the point with no clearance-volume scavenging being taken as the indicated mean effective pressure with 40° overlap at equal exhaust and inlet pressures:

Engine speed (rpm)	Inlet pressure (in. Hg abs.)	imep (lb/sq in.)	p_e/p_m	imep (lb/sq in.)	Power ratio
		130° valve overlap		40° valve overlap[a]	
2000	25	144	0.72	121	1.19
2000	35	201	.71	172	1.17
2000	45	264	.67	226	1.17
2500	25	139	.56	115	1.21
2500	35	192	.68	161	1.19
2500	45	249	.65	204	1.22
2800	30	155	.80	132	1.17
2800	40	208	.74	177	1.17

[a]For 40° valve overlap, p_e/p_m is 1.0.

Because the compression ratio was 6.1, the theoretical ratio of indicated mean effective pressure with complete clearance-volume scavenging to the indicated mean effective pressure with no scavenging should be 1.195. Because the data are quite close to this value, nearly complete clearance-volume scavenging was apparently obtained in all cases with 130° valve overlap. This same result might have been deduced from the charge-air and indicated-mean-effective-pressure curves of figure 4, where it can be seen that the charge-air flow increased about 10 percent as the values of p_e/p_m decreased beyond the point of maximum power with 130° overlap. Any residual gas in the combustion chamber should have been removed by this additional air flow with a consequent increase in power but the power did not increase, which indicated that no residual gas was left in the clearance volume, that is, complete scavenging was obtained.

The curves of figure 4 for 130° valve overlap indicated a loss in indicated mean effective pressure of about 3 percent at low p_e/p_m values. This loss was accompanied by an increase in air consumption of approximately 10 percent and as long as the air flow continued to increase the indicated mean effective pressure decreased, but when the maximum air flow was eventually reached the indicated mean effective pressure remained very nearly constant with further decrease in p_e/p_m. Thus the power variation in this region was apparently a result of the excessive scavenging air flow, which carried with it a greater percentage of the total heat of combustion to the exhaust. This heat loss overcame the increased charge-density effect expected to result from internal cooling and the net result was a slight loss rather than a gain in indicated mean effective pressure at very low values of p_e/p_m.

Specific air consumption and volumetric efficiencies. - The indicated specific air consumption and the volumetric efficiency are plotted against p_e/p_m in figure 5. In this report, volumetric efficiency is defined as the ratio of the volume of air inducted into the cylinder, determined from measurements of pressure and temperature at the inlet surge tank, to the displacement volume of the cylinder. The air inducted included the air remaining in the cylinder as well as that lost in the scavenging process. The volumetric efficiency varies over a considerably wider range with 130° valve overlap than with 40° valve overlap owing to both the large scavenging air flow and charge air wasted at low values of p_e/p_m and to backflow of exhaust gas at high values of p_e/p_m. The data at any one speed and valve overlap fell on the same curve regardless of change in inlet pressure. The maximum value of volumetric efficiency increased with a decrease in engine speed because of the increased time available for scavenging air flow.

The large increase in rate of indicated specific air consumption per indicated horsepower with decreased values of p_e/p_m is characteristic of large valve overlap. The small range of pressure ratios in which scavenging was obtained with no loss of fresh charge consists of the region between the intersection of the volumetric-efficiency curves and the intersection of the specific-air-consumption curves. This region is quite narrow especially at an engine speed of 2000 rpm showing that scavenging air was being wasted almost as soon as it began to flow. The small increase in indicated specific air consumption with 40° overlap as the value of p_e/p_m decreased was apparently caused by some change in combustion efficiency because no charge air was lost with standard valve timing.

Charge-air utilization. - The charge-air utilization is presented in figure 6 for engine operating conditions of 25 and 35 inches of mercury absolute at 2000 rpm and 25 inches of mercury absolute at 2500 rpm. From gas-sample data showing the fuel-air ratio at the time of combustion, the amount of air actually entering into the combustion process was computed by the equation

$$a = \frac{F}{(F/A)}$$

where

a charge air in cylinder at time of combustion, (lb/min)

F/A fuel-air ratio at time of combustion as obtained from gas-sampling apparatus

F fuel flow to cylinder, (lb/min)

The values of actual charge air burned are shown together with the total charge air for both the 130° and 40° overlap valve timings in figure 6. The total charge-air flow was obtained from the air-measuring system in the combustion-air line, as previously described.

The utilization factor is defined as the part of the total charge air utilized by the cylinder and is expressed by the equation

$$\text{Utilization factor} = \frac{a}{\text{Total charge-air flow}}$$

The following table shows some values of the utilization factor obtained from the curves of figure 6:

Inlet pressure (in. Hg abs.)	Engine speed (rpm)	Exhaust pressure / Inlet pressure p_e/p_m	Charge air in cylinder a (lb/min)	Total charge-air flow (lb/min)	Utilization factor
25	2000	0.9	6.32	6.50	0.972
		.7	6.85	7.70	.890
		.5	7.05	8.30	.850
		.3	7.08	8.55	.828
		.1	7.08	8.58	.825
35	2000	0.9	8.75	8.80	0.995
		.7	9.75	10.85	.900
		.5	9.80	11.80	.830
		.3	9.80	12.10	.810
		.1	9.80	12.15	.805
25	2500	0.9	7.60	7.60	1.000
		.7	8.40	8.90	.945
		.5	8.40	9.50	.885
		.3	8.40	9.75	.860
		.1	8.40	9.80	.860

The utilization factor rapidly decreased with an increase in scavenging because at the lower values of p_e/p_m a considerable amount of the increase in air flow was wasted in the scavenging process. For example, at an engine speed of 2000 rpm, an inlet pressure of 35 inches of mercury absolute and a value of p_e/p_m of 0.5, the charge air trapped in the clearance volume with 130° valve overlap was increased approximately 0.70 pound per minute over that trapped with 40° valve overlap whereas the increase in total charge air supplied was approximately 2.70 pounds per minute. The difference in these quantities decreased with increased exhaust pressure and at a value of p_e/p_m of 0.70 only 50 percent of the increased charge-air flow with 130° overlap was wasted in scavenging the clearance volume. Even before complete scavenging was obtained, a considerable amount of air was lost during the 130° overlap period. Because complete removal of residual gases from the clearance volume has been obtained on a four-valve cylinder with small differences between inlet and exhaust pressures (reference 2), there is a possibility that, up to the point of complete scavenging, less air will be wasted in the scavenging process on a four-valve engine than on a two-valve engine.

It is evident from the preceding table that a smaller percentage of charge air was wasted at 2500 rpm than at 2000 rpm. As the p_e/p_m ratio was decreased from approximately 0.5, there was comparatively little increase in percentage of scavenging air wasted at either speed. The utilization factor was approximately the same at 2000 rpm for inlet pressures of both 25 and 35 inches of mercury, which indicated that at a constant speed, the utilization factor was dependent only on variations in the value of p_e/p_m, within the range of pressures investigated.

Effect of increased valve overlap on cylinder temperatures. - The effect of increased valve overlap on the cooling of the cylinder was determined by comparing the temperatures of the exhaust-valve guide and seat for 130° valve overlap with those for 40° valve overlap as the exhaust-inlet pressure ratio was varied over a wide range at constant inlet-manifold pressure, engine speed, and rear-spark-plug-gasket temperature. A reduction of as much as 50° F in the exhaust-valve-guide temperature was obtained with the 130° overlap but no significant change was observed in the exhaust-valve-seat temperature. The increased charge air burned with the 130° overlap tended to increase these temperatures, whereas the presence of the scavenging air had a cooling effect. Because the exhaust-valve-guide temperature normally ran much hotter than the exhaust-valve seat, the scavenging air was more effective in cooling the exhaust-valve guide than the valve seat. The scavenging air was therefore effective in reducing the hottest temperature and at the same time permitted no increase in the valve-seat temperature despite the increased power output of the cylinder.

SUMMARY OF RESULTS

From an investigation of the effects of valve-overlap scavenging on the performance of an aircraft cylinder, the following results were obtained:

1. With 130° valve overlap, nearly complete clearance-volume scavenging was obtained with a consequent power increase of approximately 20 percent as compared with that obtained with 40° overlap at an exhaust-pressure-to-inlet-pressure ratio of 1.0.

2. Scavenging without loss of fresh charge was limited to a narrow range of exhaust-pressure-to-inlet-pressure ratios, especially at the lower engine speeds.

3. In order to obtain complete scavenging on the two-valve cylinder, approximately 50 percent of the increased charge air was wasted in the scavenging process.

4. Although a power increase was obtained by using 130° overlap, the cooling effect of the scavenging air was sufficient to hold the exhaust-valve-seat temperature constant and to reduce the exhaust-valve-guide temperature by as much as 50° F.

Flight Propulsion Research Laboratory,
National Advisory Committee for Aeronautics,
Cleveland, Ohio, August 8, 1947.

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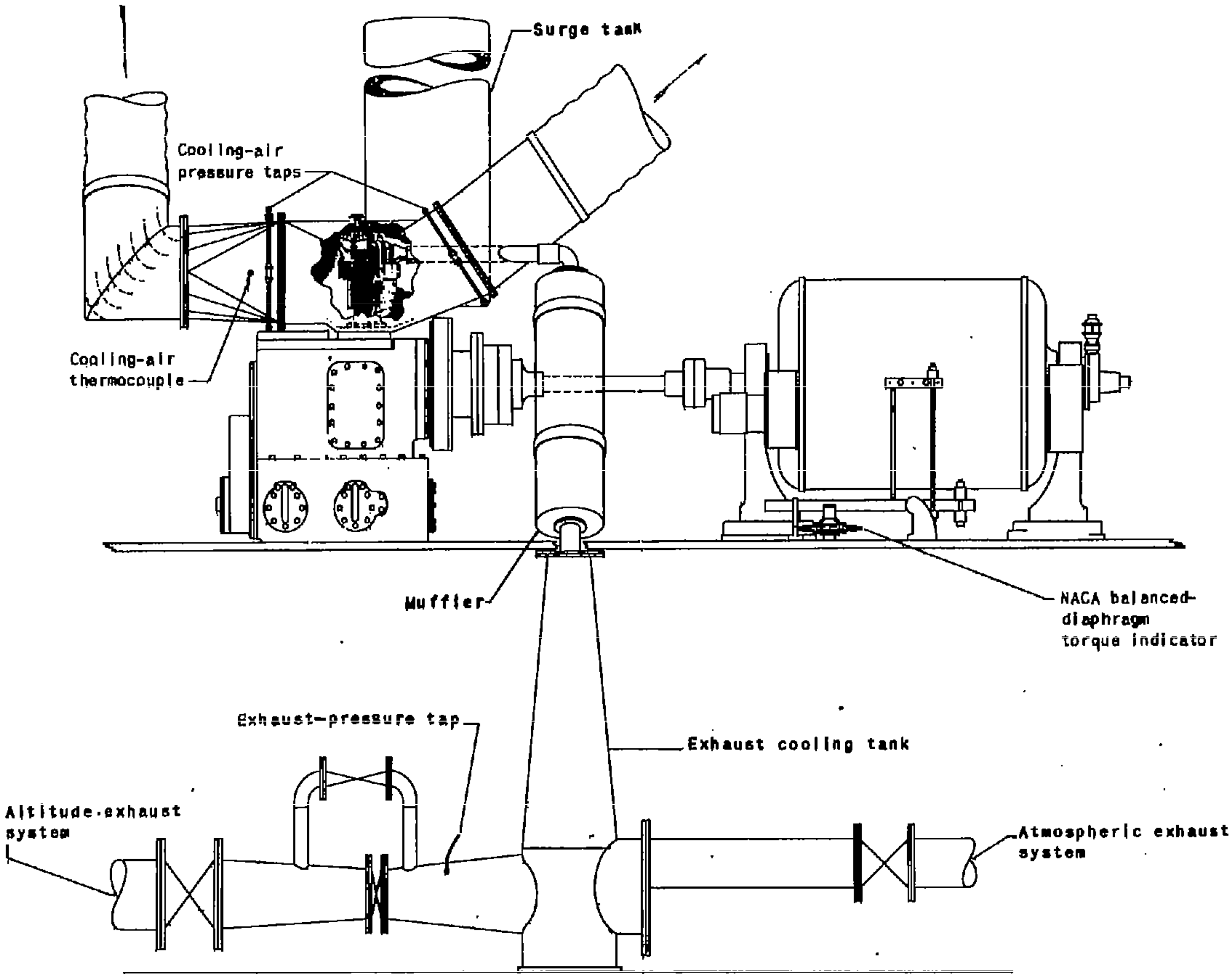

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Figure 1. - Schematic diagram of engine.

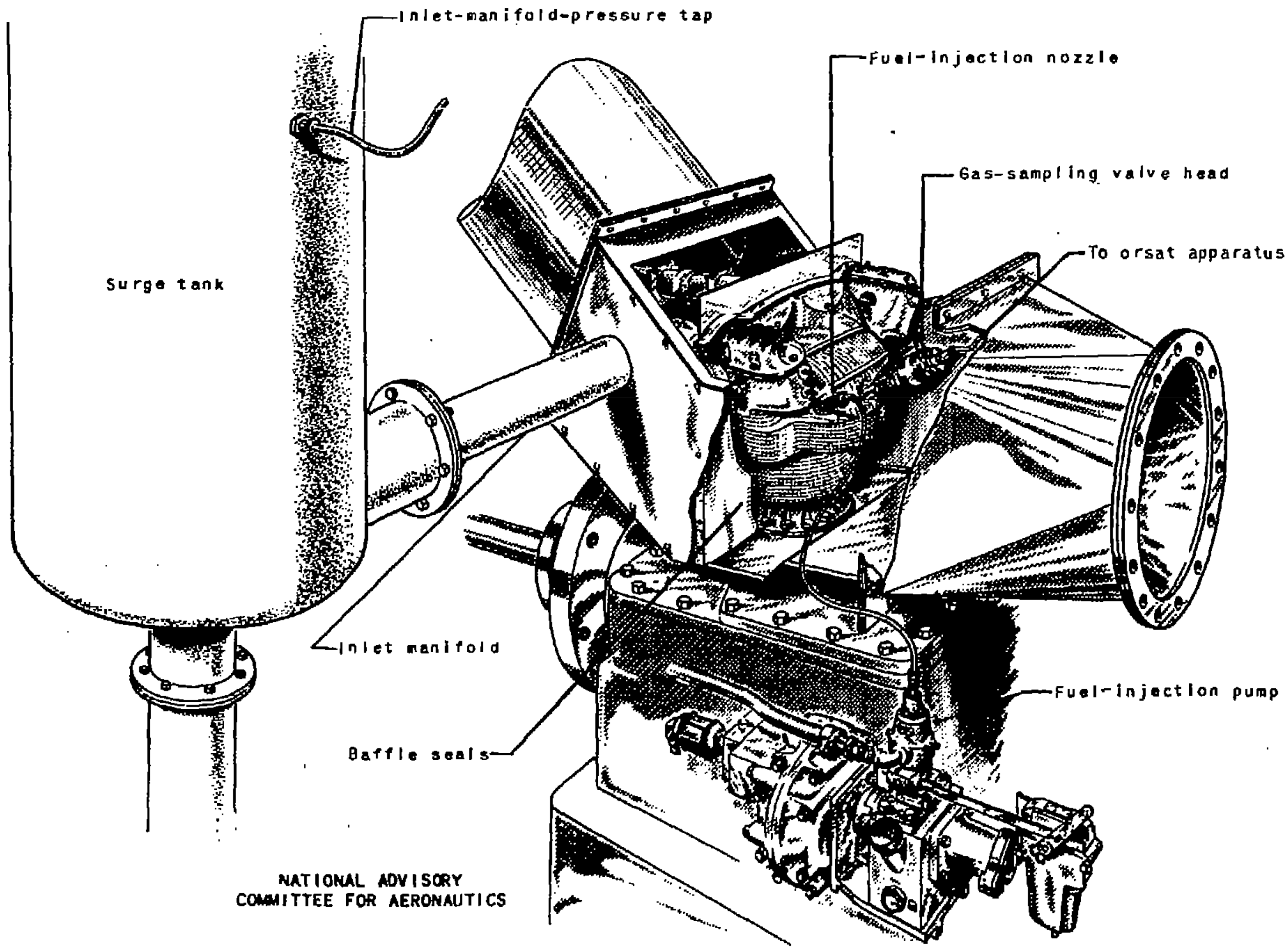

Figure 2. - Sketch of experimental section.

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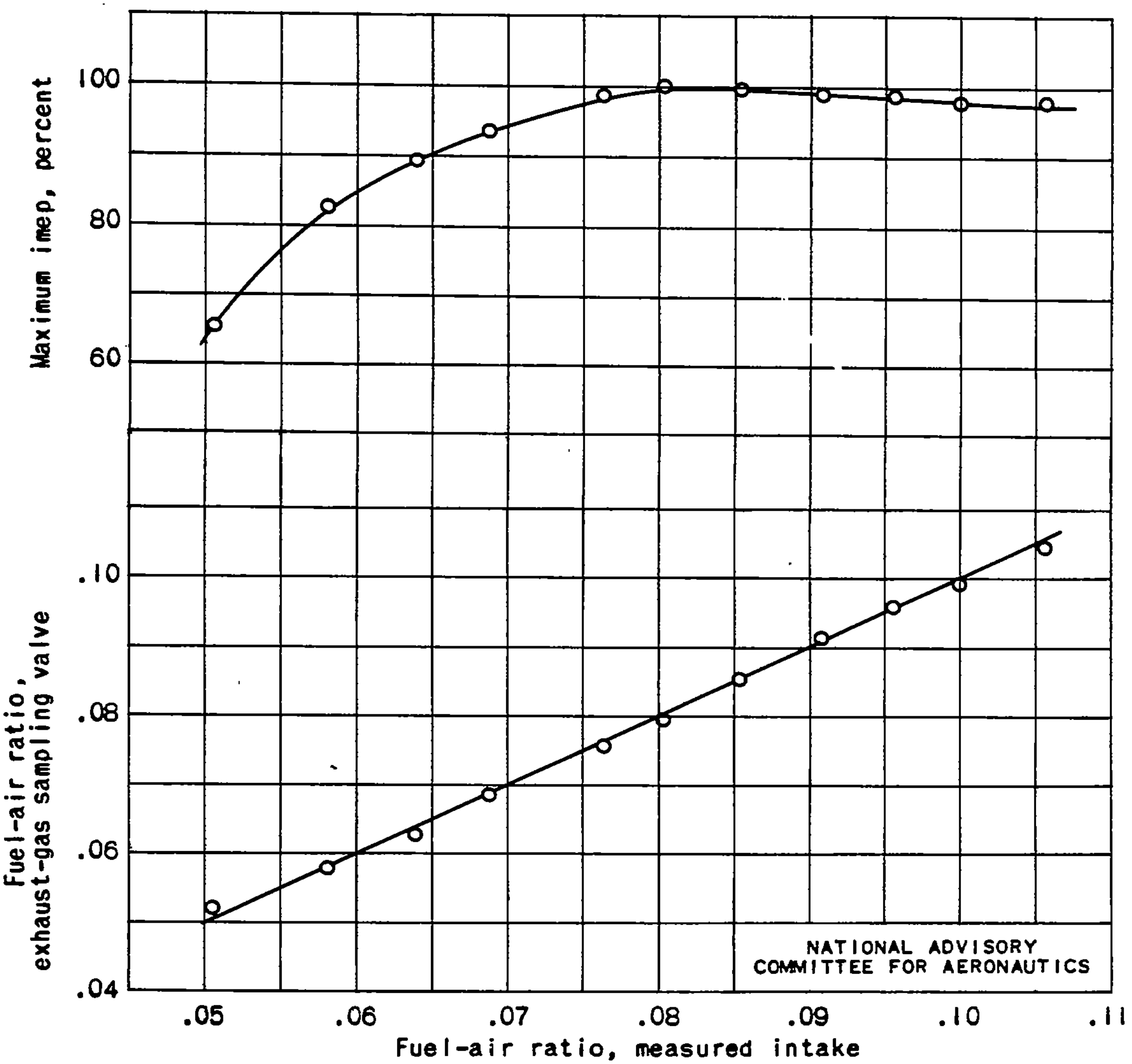
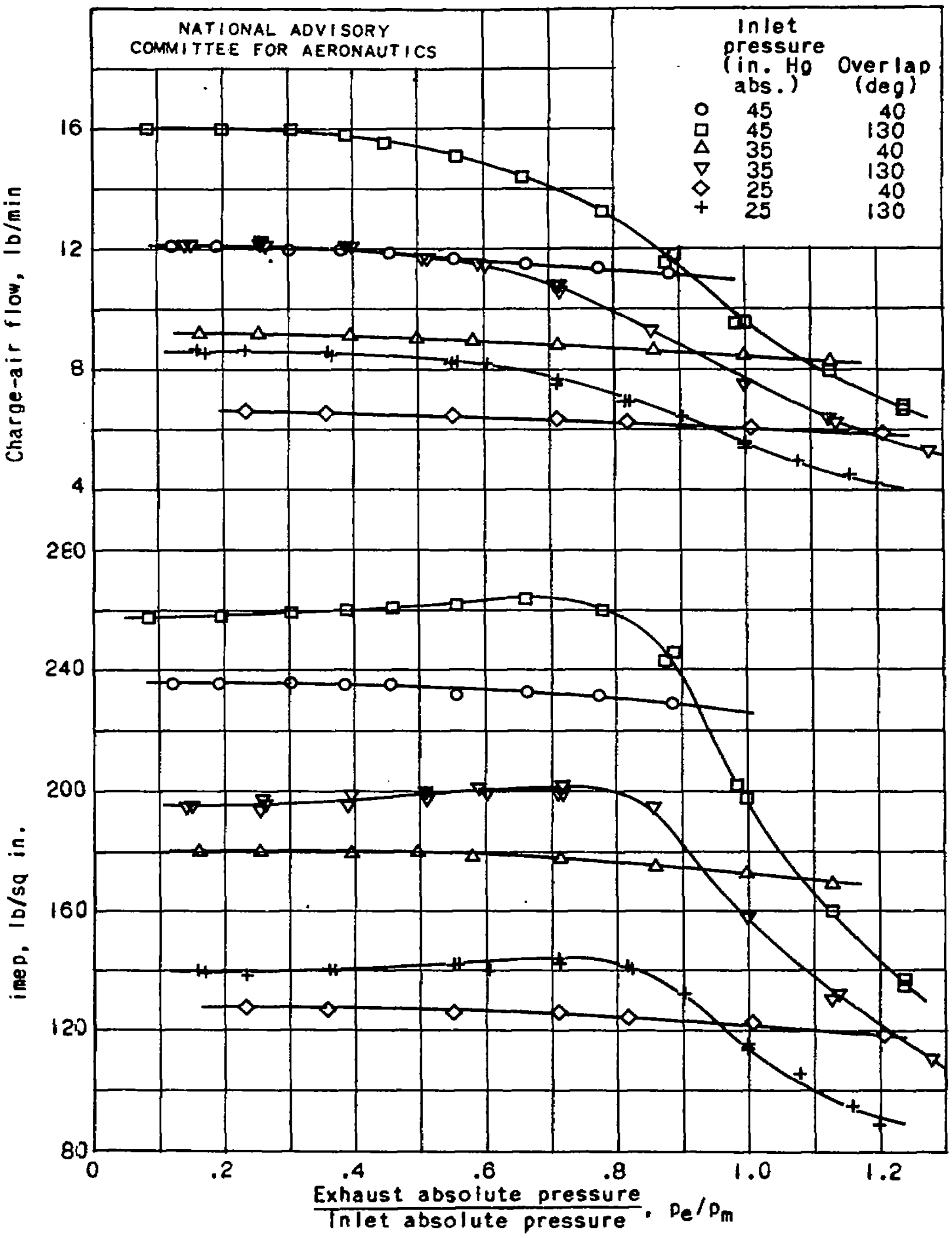


Figure 3. - Accuracy of exhaust-gas sampling valve and percentage change in power output with fuel-air ratio. Valve overlap, 40°; inlet-manifold pressure, 35 inches mercury absolute; exhaust pressure, 35 inches mercury absolute; engine speed, 2000 rpm; combustion-air temperature, 200° F.



(a) Engine speed, 2000 rpm.

Figure 4. - Variation of indicated mean effective pressure and charge-air flow with ratio of exhaust absolute pressure to inlet absolute pressure.

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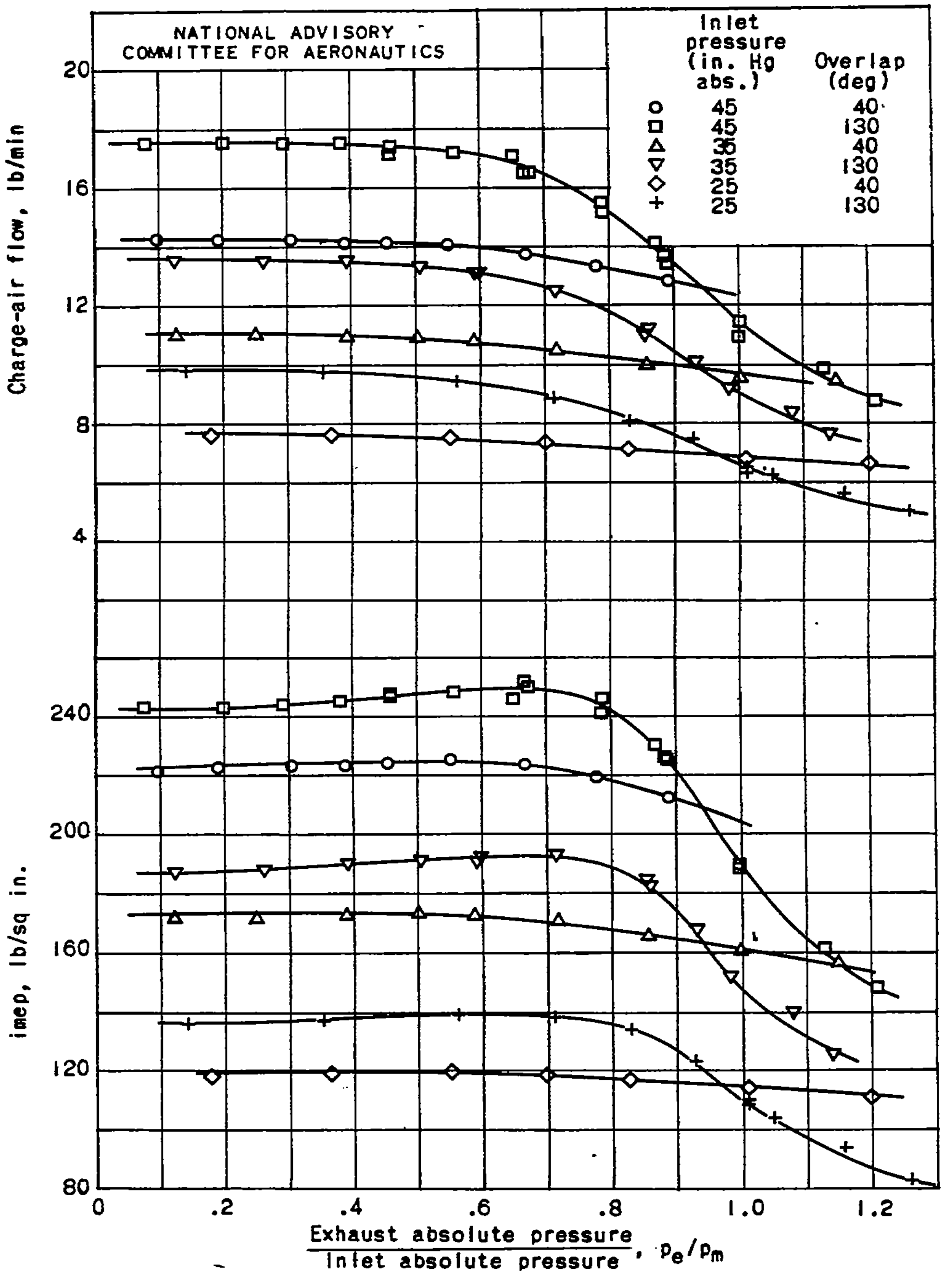
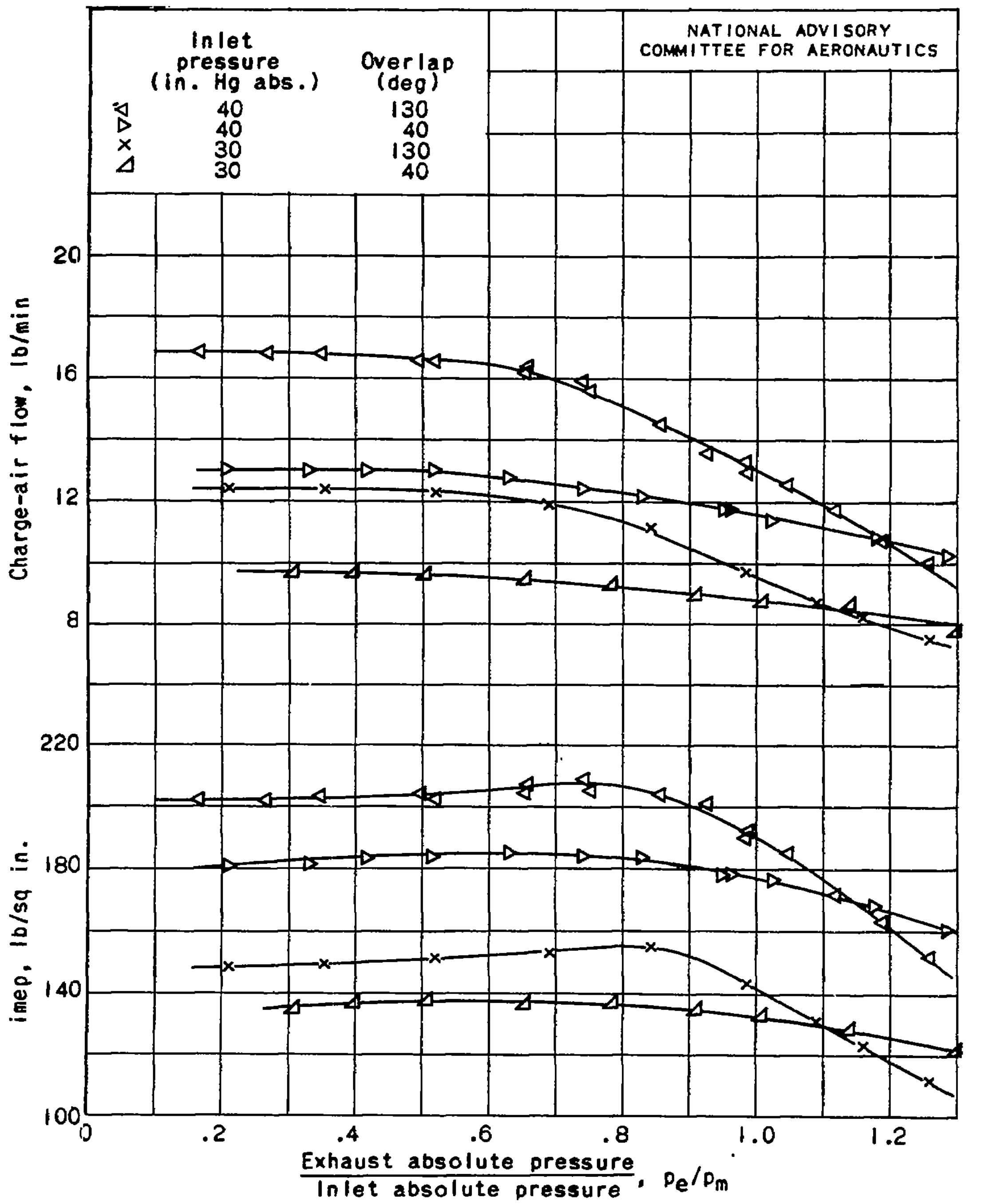


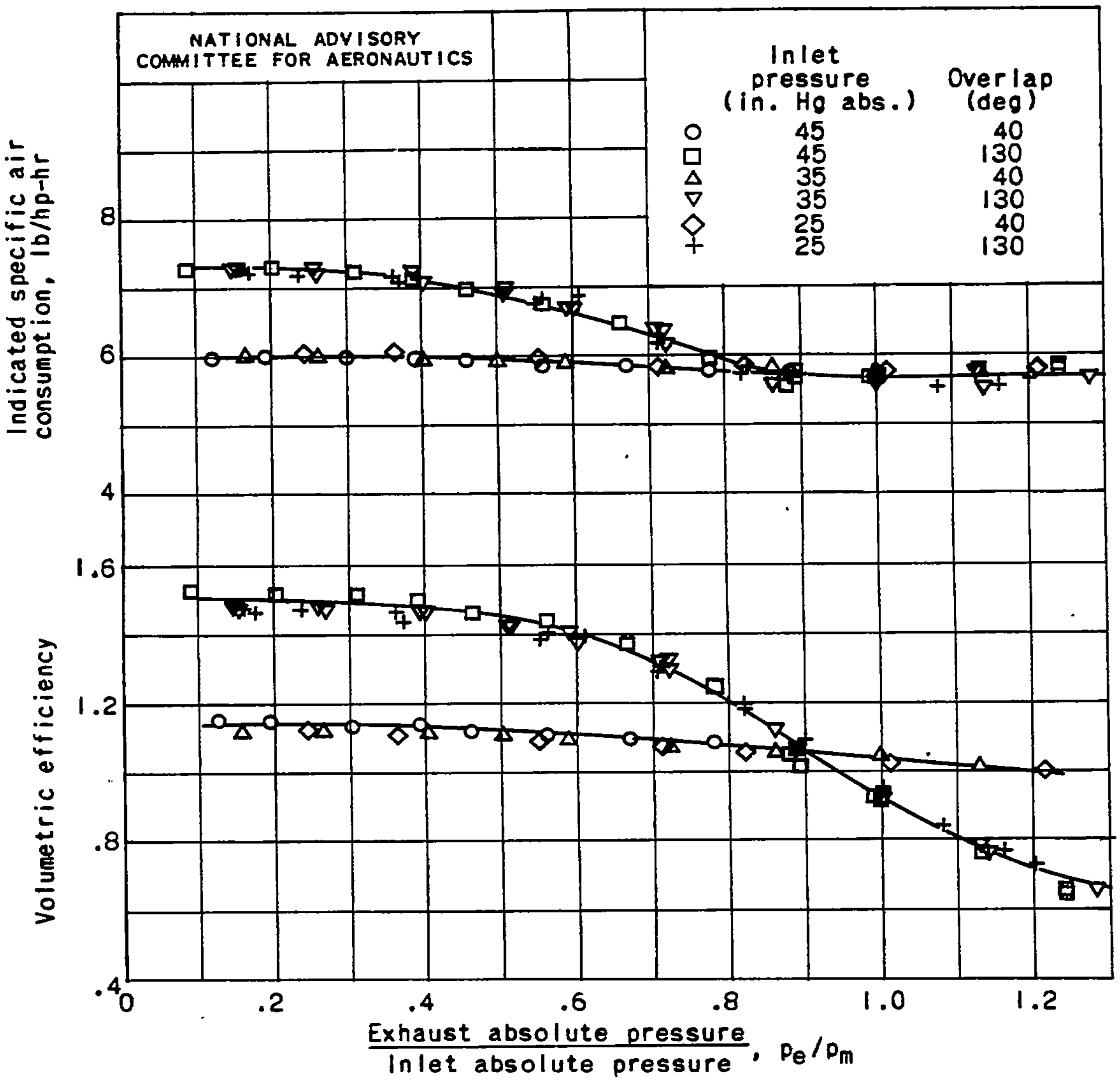
(b) Engine speed, 2500 rpm.

Figure 4. - Continued. Variation of indicated mean effective pressure and charge-air flow with ratio of exhaust absolute pressure to inlet absolute pressure.



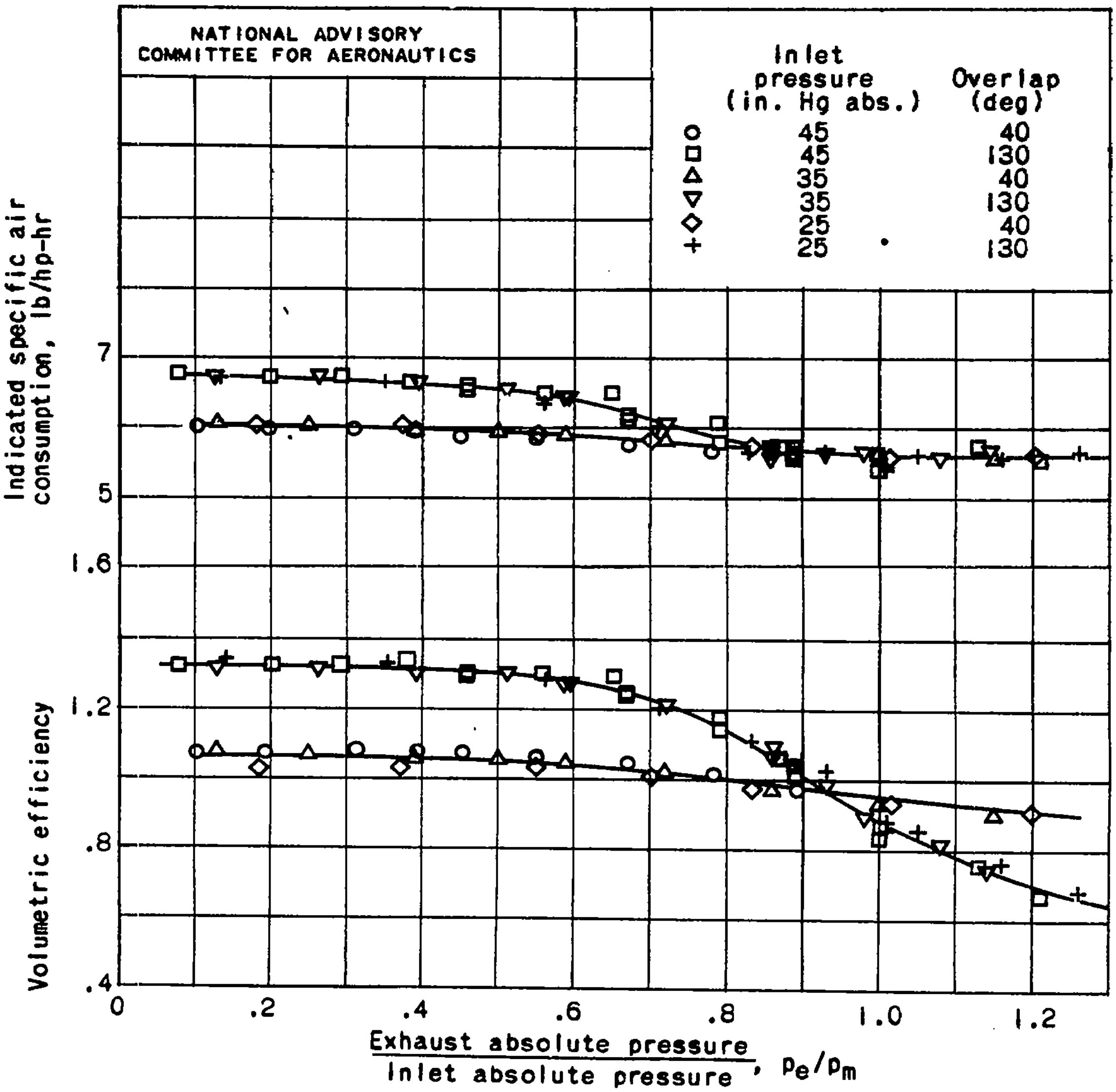
(c) Engine speed, 2800 rpm.

Figure 4. - Concluded. Variation of indicated mean effective pressure and charge-air flow with ratio of exhaust absolute pressure to inlet absolute pressure.



(a) Engine speed, 2000 rpm.

Figure 5. - Variation of volumetric efficiency and indicated specific air consumption with ratio of exhaust absolute pressure to inlet absolute pressure.



(b) Engine speed, 2500 rpm.

Figure 5. - Continued. Variation of volumetric efficiency and indicated specific air consumption with ratio of exhaust absolute pressure to inlet absolute pressure.

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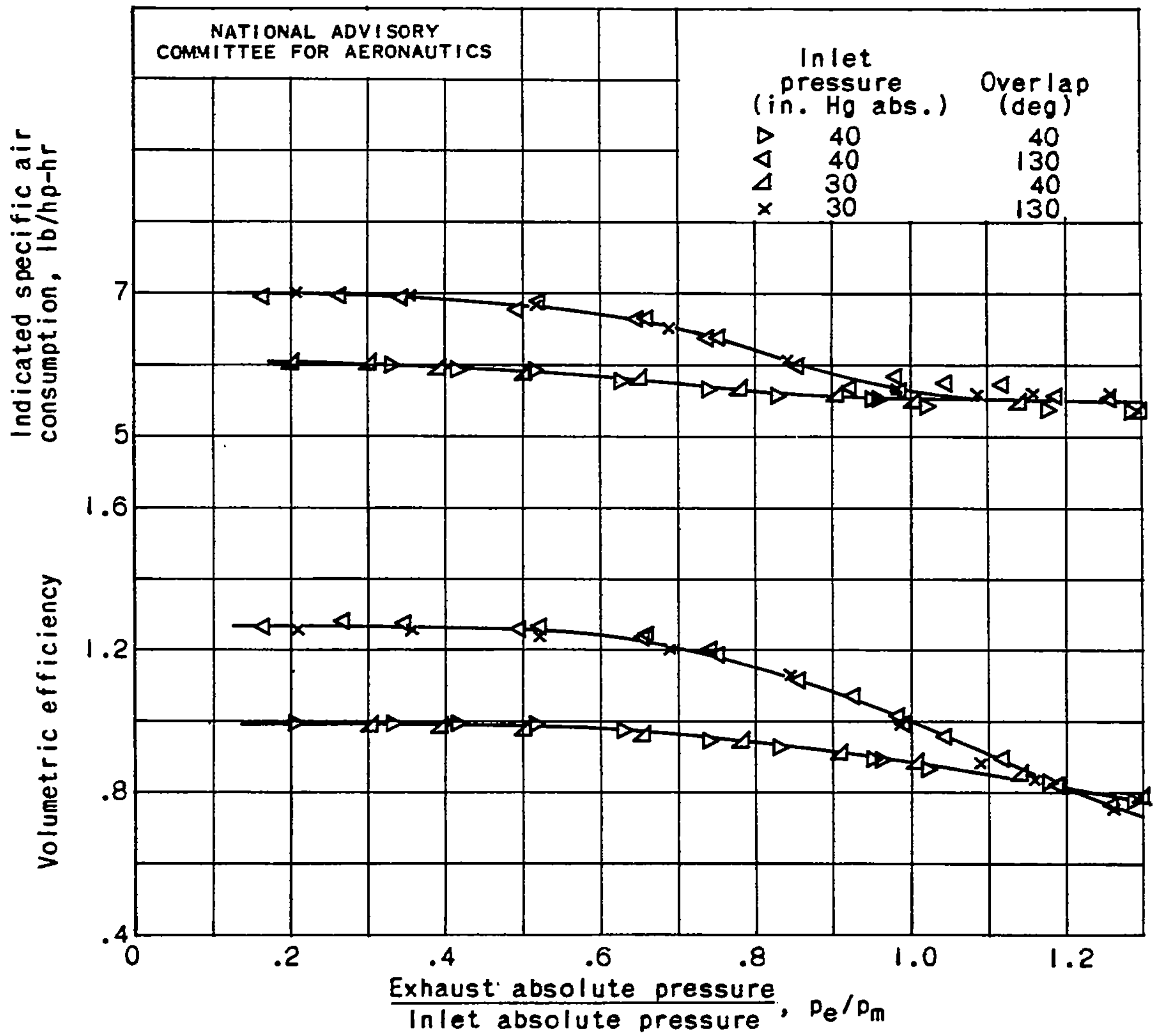

(c) Engine speed, 2800 rpm.

Figure 5. - Concluded. Variation of volumetric efficiency and indicated specific air consumption with ratio of exhaust absolute pressure to inlet absolute pressure.

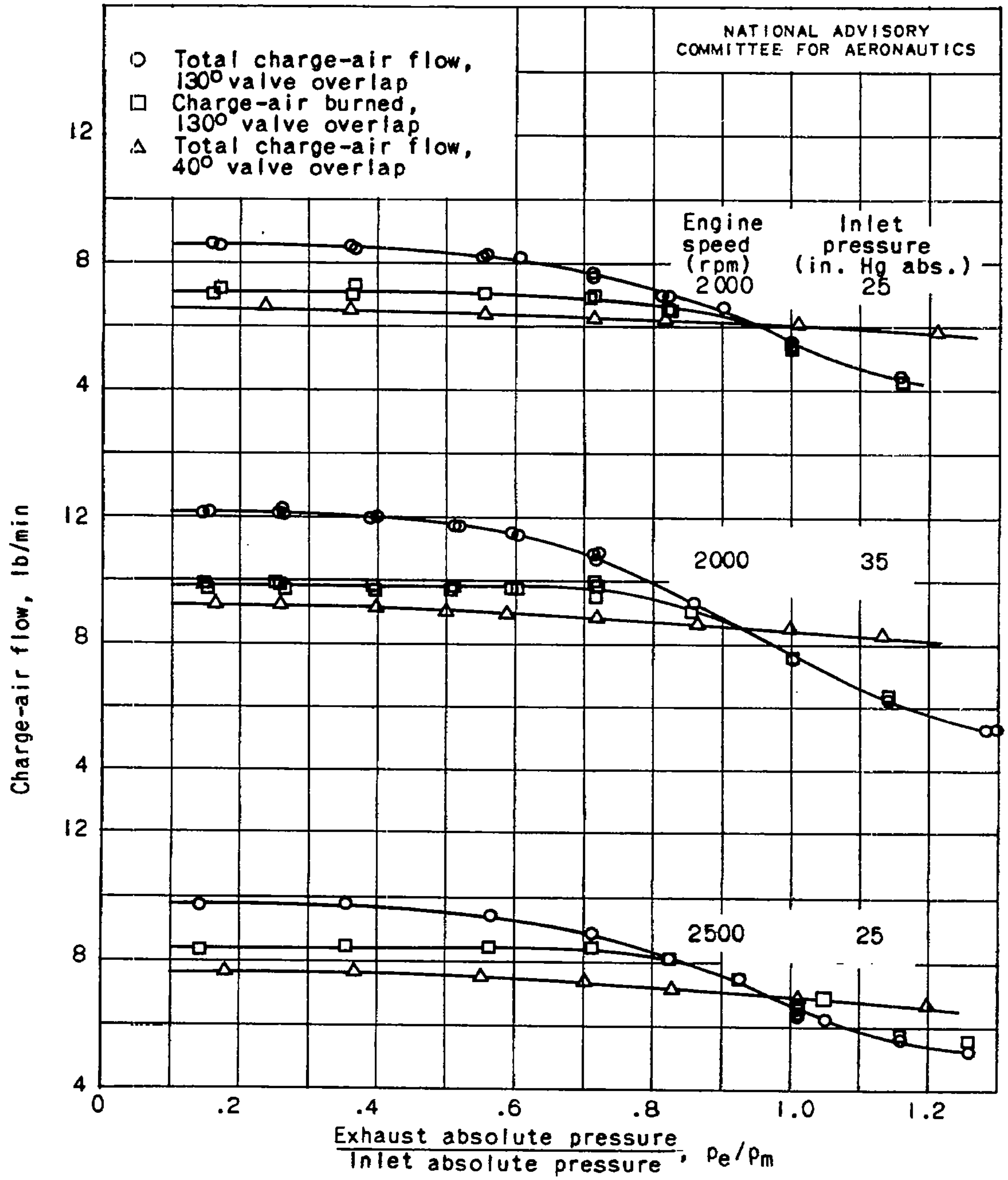

Figure 6. - Utilization of charge-air flow with 130° valve overlap determined from combustion-gas samples.

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